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INVESTIGATION OF REGENERATORS AND PULSE TUBE CRYOGENIC COOLERS

Richard A. Lechner

May 1971

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A new single stage pulse tube, with a concentric configuration, was designed and fabricated. The new design has improved detector interface characteristics and a simplified regenerator removal capability. Performance tests verified that the concentric pulse tube design is operationally feasible.

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By

RI ARD A. LECHNER

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INTRODUCTION

Regenerators

A regenerator is a very efficient compact heat exchanger which is used in cryogenic systems such as the Stirling cycle, Gifford-McMahon cycle, Solvay cycle, Vuilleumier cycle, and Pulse Tube refrigerator types. It is constructed of a matrix material that has the capability of quickly transferring and storing heat from a gas which passes through it. It is also highly resistant to heat flowing along its longitudinal direction.

As compared to the counterflow heat exchanger, the regenerator does not require simultaneous continuous flow of two physically separated fluids. The regenerator transfers heat to and from the same gas by the action of the intermediate heat transfer with the regenerator material each time the gas direction is cyclically reversed. For an equivalent thermal efficiency, the regenerator can be made much smaller and lighter than its counterflow heat exchanger counterpart.

The regenerator has several distinct advantages: it can be made relatively small; its efficiency is very high; matrix materials are readily available; its construction is simple, and, as a result, its cost is comparatively low. In normal operation, the regenerator is relatively insensitive to plugging by impurities in the gas stream. It does not require the physical separation of the gas stream.

All cryogenic refrigerators require a heat exchanger which separates the high temperature gas from the low temperature gas. Figure 1 shows the relative position of the heat exchanger in a cooling system. The use of a regenerator in this position allows a large temperature differential to be produced, with the advantages of simplicity, small physical size, and high efficiency.

The action of a regenerator can be seen as represented by Figure 2. During the steady state operation of a regenerator, for one cycle, the warm gas from a compressor at temperature T_H, passes through the regenerator, from point 1 to point 2, and progressively transfers heat to the matrix material until the gas temperature approaches the refrigeration temperature, T_R. The gas is further cooled by the refrigeration process, from point 2 to point 3, after which it returns to the regenerator. Progressively rising in temperature, from point 3 to point 4, the gas leaves the regenerator slightly colder than when it first started the cycle. The regenerator is never quite able to cool the incoming gas stream down to the refrigeration temperature, nor is it able to warm up the outgoing gas stream quite to the incoming gas temperature, because of heat losses and irreversibilities. The lower the refrigeration temperatures, the greater the effect of the losses. This makes heat exchanger efficiencies of 97%, and higher, a necessity for most cryogenic applications.

Losses in regenerators have been categorized and described (References 1 and 2). These effects, such as property variations with temperature, longitudinal heat conduction, wall effect, feed gas flow effect, and end effects, all combine to provide the total heat loss which reduces the regenerator's overall efficiency.

Various types of matrix materials can be used in a regenerator, according to the characteristics desired, but certain properties are required. Resistance to gas flow should be low to minimize the energy required to move the gas through the regenerator. The construction should provide a large heat transfer area to enable the gas to experience intimate contact with the matrix material. The heat capacity and thermal conductivity should be large to enable rapid transfer of large amounts of heat.

It has been common practice in the study of regenerator losses to determine regenerator inefficiency, rather than efficiency (Reference 3). The inefficiency is defined as:

$$I_e = \frac{\Delta T_A}{T_H - T_R}$$

Where ΔT_A = Average Temperature Difference at the Cold End

T_H = Warm End Temperature

T_R = Cold End Temperature

The inefficiency represents the ratio of heat that is not transferred, to the maximum heat that can be transferred. The regenerator total thermal loss, $\dot{Q}_L$, is given by the product of I_e and the maximum heat that can be transferred. That is:

$$\dot{Q}_L = I_e \dot{m} c_p (T_H - T_R)$$

Where $\dot{m}$ = Gas Flow Rate

c_p = Specific Heat of Gas

In a regenerator, the outlet gas temperature varies through a large range and in a nonlinear manner. Figure 3 shows a typical outlet gas temperature variation with time, at the cold end, during a cooling pulse period, ϕ_c. Considering that ϕ_c may be a period of only 0.1 to 0.5 second, the determination of a good experimental temperature average would require many readings during a pulse period and, therefore, be very difficult to obtain.

The inefficiency, however, can be evaluated through testing methods by measuring Q_L, from a liquid nitrogen boil-off rate (Reference 3). The gas flow rate can be measured by using a calibrated orifice, a manometer, and a pressure gage. T_R, T_H and c_p are essentially constant. Hence, the inefficiency can be calculated by:

$$I_e = \frac{\dot{Q}_L}{\dot{m}\, c_p\, (T_H - T_R)}$$

Regenerator inefficiencies are expected to vary in the range of 0.5% to 3.0% (Reference 4).

Pulse Tube Refrigeration

Pulse tube refrigeration has been shown (References 5 and 6) to be a unique method of obtaining cryogenic temperatures. A pulse tube makes refrigeration available through a heat pumping mechanism which pumps heat up along the walls of the tube from the low temperature end to the warm end thermal reservoir. Physically, as shown in Figure 4, a pulse tube simply consists of a regenerator, a high and low temperature heat exchanger separated by a hollow tube, a compressor, a flow reversing valve, and a porous plug to produce smooth, slug like, flow.

The physical interpretation of the heat pumping mechanism is described when an element of gas, as shown in Figure 4, is traced through its cyclic processes. The gas coming from the compressor passes through the flow reversing valve and regenerator and into the pulse tube. The gas is compressed from a lower pressure, P_L, at point 1, to a higher pressure, P_H, and higher temperature at point 2. Each small particle of gas during a small finite period of time gives up heat, from point 2 to point 3, at constant pressure, to the wall of the pulse tube. The pressure is released, by the action of the flow reversing valve, and the gas then moves from P_H at point 3 to P_L at point 4. The gas at point 4 is then at a slightly lower temperature than at point 1. At point 4 the gas receives heat from the pulse tube wall, at constant pressure, increasing its temperature up to point 1, completing the cycle. This action produces a heat pumping action along the length of the pulse tube wall. The tube then is capable of providing a hot end and a cold end with a wall temperature distribution, T_W along its length.

Although the efficiency of the pulse tube is not as high (Reference 5) as several other type cryogenic methods, the low vibration, ease of service, low noise, low speed, low cost, high reliability, and long life potential justify its investigation for military applications. Additionally, there are no low temperature moving parts in this type of refrigerator.

SECTION II

REGENERATOR TEST STATION

A Regenerator Test Station was designed and fabricated during FY-69 (Reference 5). During FY-70, the test station was modified and tested to make it operational. The Regenerator Test Station was built to generate engineering data to be used in the design of regenerators for Pulse Tube and Gifford-McMahon cycle cryogenic coolers. The test station is designed so that it can also be used in the study of regenerators for other types of cryogenic coolers, at liquid nitrogen and liquid helium temperatures.

Description of Equipment

Single stage cryogenic refrigerators usually contain only one regenerator. In order to maintain cyclic flow and be able to make measurements of the flow conditions, two regenerators are required in the Regenerator Test Station, in conjunction with a fixed temperature heat exchanger and flow reversing valve, as a means of simulating the same type of conditions that a regenerator experiences in a cryogenic refrigerator.

Figure 5 shows the overall test station as it was first assembled at the end of FY-69. Figure 6 shows the construction within the vacuum enclosure. As shown in Figure 7, helium gas is circulated by a compressor, through a heat exchanger to remove the heat of compression, an oil separator for oil removal, and a small molecular sieve filter. The small molecular sieve filter, assembled in a transparent container, removes moisture and also visually aids in determining whether the oil is, in fact, being removed by the oil separator. The gas continues on to the rotating valve where flow reversal of the gas takes place through the regenerators. Gas flows in one direction, as shown in **Figure 7**, below the valve during one half of the cycle, then reverses direction during the second half of the cycle. The gas then continues past the valve through the orifice, where its pressure is measured by a pressure gage and manometer, and returns to the compressor. Below the rotating valve, the gas passes through one of the first stage regenerators, the first stage liquid nitrogen heat exchanger, the first of the second stage regenerators, and the second stage liquid helium heat exchanger. The gas continues back to the flow reversing valve through the second of the second stage regenerators, the first stage liquid nitrogen heat exchanger, and the second of the first stage regenerators.

The heat transferred through the second stage liquid helium heat exchanger boils off liquid helium which then passes as a gas through a heat exchanger, bringing the gas to room temperature, and through a gas flow meter where the boil-off rate is measured. A vacuum is produced in the vacuum enclosure by a mechanical pump and diffusion pump. The vacuum is maintained as an insulation against heat transfer from exterior surroundings to the refrigerated heat exchangers. Superinsulation was also used to assist in reducing radiation heat transfer.

By removing the first stage regenerators, not using liquid nitrogen in the first stage dewar, and replacing the liquid helium with liquid nitrogen, regenerators can be tested in the second stage at liquid nitrogen temperatures. Using the Regenerator Test Station in this manner, sets of 3/4" diameter by 3" long regenerators were tested at 77°K.

A vacuum pump was used to evacuate the compressor system during the normal helium gas purging and charging procedures. It was discovered that the manometer fluid (water, colored by Merium Instrument Company Indicating Fluid Concentrate No. D 2930) was being evaporated by the vacuum pump. Evaporation of the fluid could not be tolerated because of moisture contamination of the regenerators. The fluid was then changed to Merium Instrument Company # D 2969 Red Unity Oil, which has the same specific gravity as water. The vacuum pump did not cause evaporation of the red unity oil.

When the lead ball test regenerators were removed for modifications, it was discovered that the screens used as retaining devices for the lead balls were severely corroded. Apparently, the evaporation of the original manometer fluid had contaminated the system with moisture and caused the corrosion discovered on the phosphor bronze screens. The corrosion that occurred is shown in the comparison photograph, Figure 8. The operation of the system prior to discovering this problem indicated a gradual plugging of the regenerators during the series of tests.

Another difficulty encountered was the choice of material for the flow reversing valve. Several plastic materials were used and found to be unsatisfactory. Graphite impregnated polyimide was finally found to provide satisfactory results as a valve material.

Unbalanced Flow

A problem which continuously created concern was a condition known as unbalanced flow. As shown in Figure 9, the portion of the system below the flow reversing valve consists of a heat exchanger (Qex), two regenerators (A and B), and two volumes (V_1 and V_2). Volumes V_1 and V_2 are shown to represent any differences in the volumes of the piping, etc. between the regenerators and the flow reversing valve. Several assumptions are made with respect to the normal operational test conditions:

1. Flow period 1 equals flow period 2, as a result of the constant speed flow reversing valve.

2. The pressure drop through V_1 and through V_2 is considered small as compared with the combination of the regenerator and heat exchanger pressure drops.

3. The regenerators are considered as having equal pressure drops.

4. The flow in one direction, through regenerator A, equals the flow through regenerator B.

As a result of the constant speed of the flow reversing valve, gas flows into the system in one direction, then is reversed and flows in the opposite direction. During the first half cycle, gas flows into V_1, where it reaches some pressure P_1 and flows through the regenerators and heat exchanger, and returns through V_2 to the flow reversing valve. Similarly, the second half cycle results in gas flowing into V_2 where it reaches pressure P_2, as it continues to flow through the regenerators and heat exchanger, through V_1, back to the flow reversing valve.

Although the regenerators and heat exchanger have the same pressure drop in both steady state gas flow directions, the cyclic gas flow will not be the same in both directions, as a result of the difference in V_1 and V_2. Assuming that the pressure P_1 is the same pressure as P_2 during each half of the flow cycle, the time required to reach that pressure will be longer for volume V_1 than for volume V_2. In this event, a gas particle moving in the first flow direction will not travel as far as it would in the second flow direction. This results in an uneven cyclic gas flow through the regenerators, and a general unidirectional gas migration is superimposed on the cyclic gas flow.

Figure 10 is representative of the gas pressure variations versus time in a balanced gas flow condition. During one valve rotation, period A represents the flow pulsation in the first flow direction, and period B represents the flow pulsation in the second direction. Figure 11 is representative of the gas pressure variation in an unbalanced flow condition. A condition of this type was serious enough to be visually observed on the Regenerator Test Station manometer.

Regenerators maintain temperature differentials uniquely on cyclic gas flow, but do not on steady state flow. With even cyclic gas flow (the same flow in each direction), regenerators maintain large temperature differentials at very high efficiencies. During steady state flow, they maintain no temperature differential. The small migration of gas, as a result of unbalanced flow, is equivalent to a small amount of steady state flow. The result of this migratory flow affects the regenerator efficiency by reducing the temperature differential that is established by the cyclic portion of the gas flow.

The Regenerator Test Station experienced the unbalanced flow condition continuously. Attempts were made to eliminate the condition, but it was impossible to do so without major changes. However, it was possible to minimize the effect to the extent that it is believed the error introduced by unbalanced flow was very small.

Regenerator Tests

The regenerator inefficiency, I_e, is defined as the ratio of the quantity of heat that is not transferred to the regenerator, to the maximum quantity of heat that can be transferred. That is, $I_e = Q_L/Q_T$, where Q_L represents the quantity of heat not transferred, and Q_T represents the maximum quantity of heat that can be transferred.

The quantity of heat not transferred, Q_L, results in a boil-off of liquid nitrogen in the Regenerator Test Station. The boil-off rate is measured and used in the calculation of Q_L as follows:

$$Q_L = (H_v)(D_n)(F)(B_o - B_L) \quad \text{where:}$$

H_v = Nitrogen Heat of Vaporization (BTU/lb)

D_n = Nitrogen Density at Standard Conditions (lb/ft^3)

F = Conversion Factor (ft^3/Liter)

B_o = Operating Boil-Off Rate (Liters/min)

B_L = Non-Operating Boil-Off Rate (Liters/min)

The maximum quantity of heat, Q_T, that can be transferred is a function of the gas flow through the regenerators and is measured by a manometer, an orifice, and a pressure gage. It is calculated as follows:

$$Q_T = (C)\sqrt{(64.4)(P_m/D)}\,(D)(T_{RT} - 139)$$

$$D = P_o\,(144)/T_{RT})\,(R)$$

$$C = 60\,(C_D)(A)(c_p)$$

A = Orifice Area (ft^2)

C_D = Orifice Constant

R = Gas Constant (Helium)

T_{RT} = Room Temperature (°R)

P_o = Orifice Pressure (Psia)

P_m = Manometer Pressure (Psia)

c_p = 1.25 (BTU/lb-m °R)

$24.4 = 28$

139 = Boiling Point of Liquid Nitrogen (°R)

60 = min/hr

$144 = in^2/ft^2$

A computer program, as shown in Appendix I, was written to assist in computing the Regenerator Inefficiency and Standard Flow Rate from the data obtained during testing.

The regenerators were tested under the following conditions to determine that the Regenerator Test Station was operating properly, and producing consistent and accurate results:

1. The regenerators were operated with end temperatures of 300° Kelvin and 77° Kelvin.

2. A constant regenerator size of three quarters of an inch diameter, and three inches long was used.

3. The regenerator matrix material used was 150, 200, and 325 mesh phosphor bronze screen; and .006 to .010 inch diameter lead balls.

4. The helium flow rate was varied between 3 to 9 SCFM.

5. Regenerator cycle rate was varied from 20 to 120 cycles per minute.

6. The regenerator wall material was phenolic.

Lead ball regenerators were fabricated and tested but did not produce consistent results. They were found to compact to a smaller volume, as a result of the flow reversal of gas through them, and the resulting compacted slug cyclicly shifted to produce audible noise. This problem could not be readily corrected, therefore, no further tests were performed on lead ball regenerators.

Sets of 150 mesh, 200 mesh, and 325 mesh phosphor bronze screen regenerators were tested. The tests conducted on these regenerators are listed as follows:

Set 1: Manometer settings at 5, 10, 15, 20, 25, and 30 inches H_2O.

Test 1. Constant valve speed of 20 RPM.

Test 2. Constant valve speed of 40 RPM.

Test 3. Constant valve speed of 60 RPM.

Test 4. Constant valve speed of 80 RPM.

Test 5. Constant valve speed of 100 RPM.

Test 6. Constant valve speed of 120 RPM.

Set 2: Valve speed settings of 20, 40, 60, 80, 100, and 120 RPM.

Test 1. Constant manometer setting of 10" H_2O - Two Runs.

Test 2. Constant manometer setting of 20" H_2O - Two Runs.

Test 3. Constant manometer setting of 30" H_2O - Two Runs.

Figures 12 through 17 show the inefficiencies of several test regenerators plotted versus valve speed (RPM) and standard cubic feet per minute (SCFM) flow rates of helium. Note that all the regenerator inefficiencies are lower than 3%.

Regenerator Pressure Drops

The data taken during the regenerator testing included the measurement of upstream and downstream pressures. The difference of these pressures represents the pressure drop across that part of the system which includes the regenerators, heat exchangers, flow reversing valve, and associated tubulation. Duplicate test runs were made at room temperature conditions, with the regenerators removed, so that the upstream and downstream pressures could again be measured. This difference in pressure represents the pressure drop of the system containing everything mentioned above except the heat exchangers. The pressure drop from the original data, less the pressure drop of the duplicated runs containing no regenerators, was considered a good approximation of the average operating pressure drop of the test regenerators. The upstream pressure ranged from 50 to 100 pounds per square inch gage (psig) during these tests. Table I is a summary of the operating pressure drops which were averaged from all the data taken during testing.

TABLE I

REGENERATOR PRESSURE DROPS

FLOW RATES (SCFM)	TOTAL REGENERATOR PRESSURE DROPS*(psi)
150 MESH PHOSPHOR BRONZE SCREEN - 0.0026" DIAMETER WIRE (490 SCREENS PER REGENERATOR)	
3.55	0.80
4.78	1.03
5.88	1.13
6.30	1.20
7.65	1.33
8.39	1.37
200 MESH PHOSPHOR BRONZE SCREEN - 0.0021" DIAMETER WIRE (670 SCREENS PER REGENERATOR)	
3.81	2.00
4.82	2.13
5.90	2.63
6.83	2.70
7.67	2.80
8.42	3.17
325 MESH PHOSPHOR BRONZE SCREEN - 0.0014" DIAMETER WIRE (1010 SCREENS PER REGENERATOR)	
3.[illegible]	4.70
4.91	7.13
6.02	8.63
6.[illegible]	9.00
7.80	10.13
8.5[illegible]	10.67

* The pressure drops are shown for the combination of two 3/4" diameter by 3" long regenerators. The pressure drop per regenerator would be half of the values tabulated.

SECTION III

PULSE TUBES

During this fiscal year a new pulse tube configuration, designed in August of 1968 (Reference 7), was fabricated and tested. The new concentric configuration has the advantages of better interface characteristics and the ease of changing regenerators. This latter characteristic allowed tests to be run using different types and lengths of regenerators.

Three models of the concentric pulse tube were constructed. A 1" diameter by 8" long tube, a 1" diameter by 10" long tube, and a 1/2" diameter by 10" long tube were fabricated. The 1" diameter by 8" long tube was the first model constructed. It was the only tube of the three constructed that contained a sintered metal flow straightener. Its performance tests resulted in consistent bottom temperatures of 135°K. Figure 18 shows an exploded view of the first 1" diameter by 8" long concentric pulse tube. Figure 19 shows the assembled 8" long pulse tube attached to the rotary valve.

The 1" diameter by 10" long pulse tube was constructed similar to the 8" long tube, with some minor changes. The inner tip was fabricated of stainless steel instead of copper, and the inner tube assembly was designed to be removable for the purpose of testing the effect of the volume between the inner and outer tubes. By methodical testing, which included changing the regenerator matrix material and length, and by varying the valve speed and operating pressures, the 10" pulse tube was able to achieve a no load temperature of 160°K. This was done by operating at 190 psig high pressure, 33 psig low pressure, and 82 RPM valve speed. The cool down was accomplished in approximately 36 minutes.

The 1/2" diameter by 10" long tube was constructed similar to the 1" diameter by 10" long tube. Difficulties with fabrication of the regenerator rendered the few resulting tests unsatisfactory.

SECTION IV

CONCLUSIONS

1. A Regenerator Test Station, capable of testing regenerators at liquid nitrogen and liquid helium temperatures, was completed and made operational. Limited testing of several regenerators at 77°K, conducted to check the accuracy of the test station, resulted in regenerator inefficiencies of less than 3%, which is in the expected range and compared closely to published data.

2. The unbalanced flow condition was minimized to a relatively small error, however, it could not be completely eliminated without major changes in the test station.

3. The Regenerator Test Station provides a means for testing regenerators and for generating design data on regenerators of various lengths, diameters, and matrix materials, and at two temperature levels, namely, 77°K and 4.2°K. This data will be generated and used in the future in the design of regenerators for cryogenic refrigerators.

4. Several concentric pulse tubes were designed, fabricated, and tested. A single stage 1" diameter by 10" long tube achieved a no-load temperature of 160°K. The concentric pulse tube has two distinct advantages. First, its simple cylindrical shape reduces interface problems with electronic devices, such as infrared detectors. Second, the regenerator section, located in the center of the tube, can be made removable for ease of maintenance, repair, or replacement.

5. Testing of the concentric pulse tube indicated that its operation was not dependent upon a flow straightener. With or without the flow straightener, the performance of the tube was the same.

6. With the capability of easily changing regenerators in the concentric pulse tube, it was found, as expected, that the refrigeration temperature was drastically affected by the type and size of the regenerator. This emphasizes the need and importance of regenerator testing.

7. The investigation of the pulse tube will be continued since it has the potential to be a very low cost and highly reliable cooler. The Far Infrared Technical Area of the Night Vision Laboratory of USAECOM has a strong interest in the further development of the pulse tube for applications such as cooling detector arrays in airborne far infrared imaging systems.

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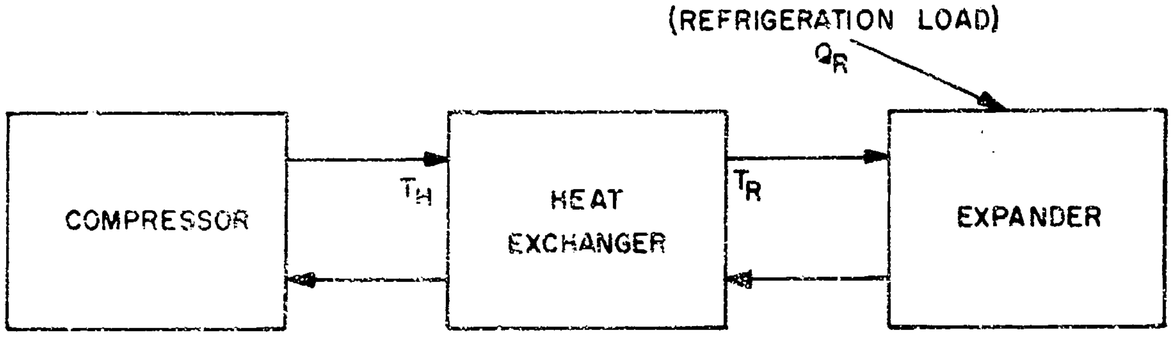

FIG. 1 COOLER SYSTEM DIAGRAM

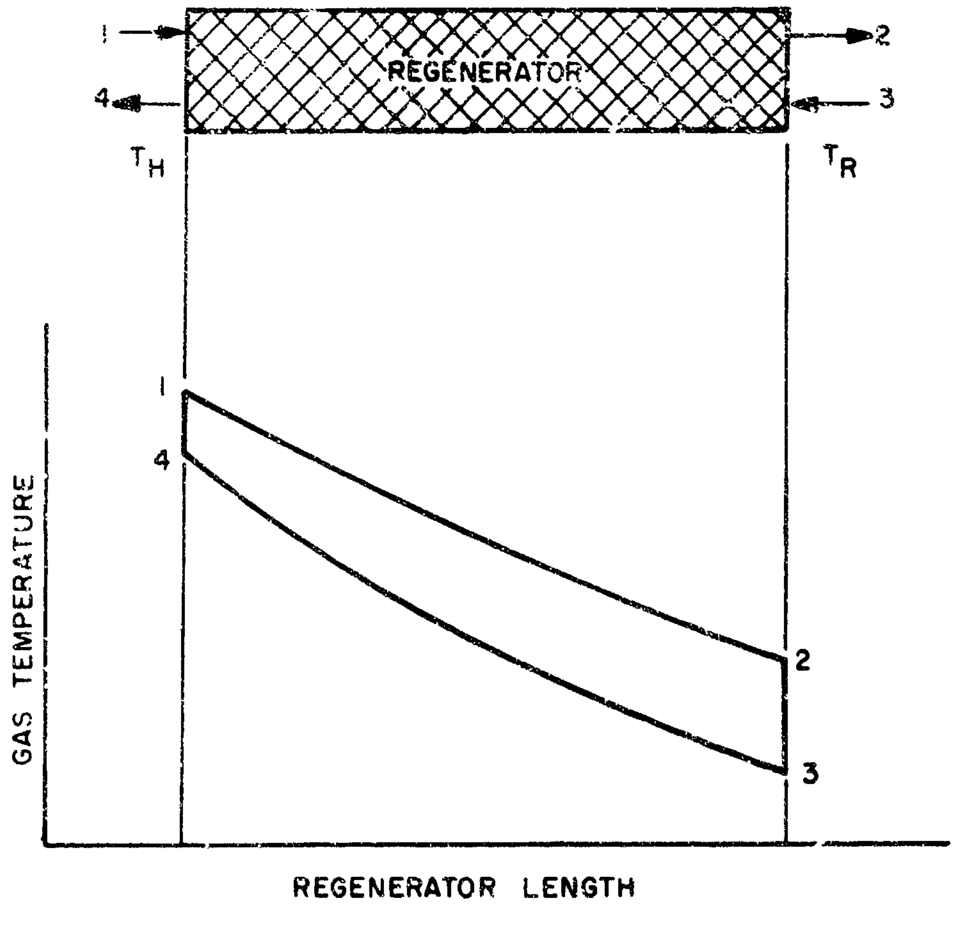

FIG. 2 REGENERATOR TEMPERATURE CYCLE

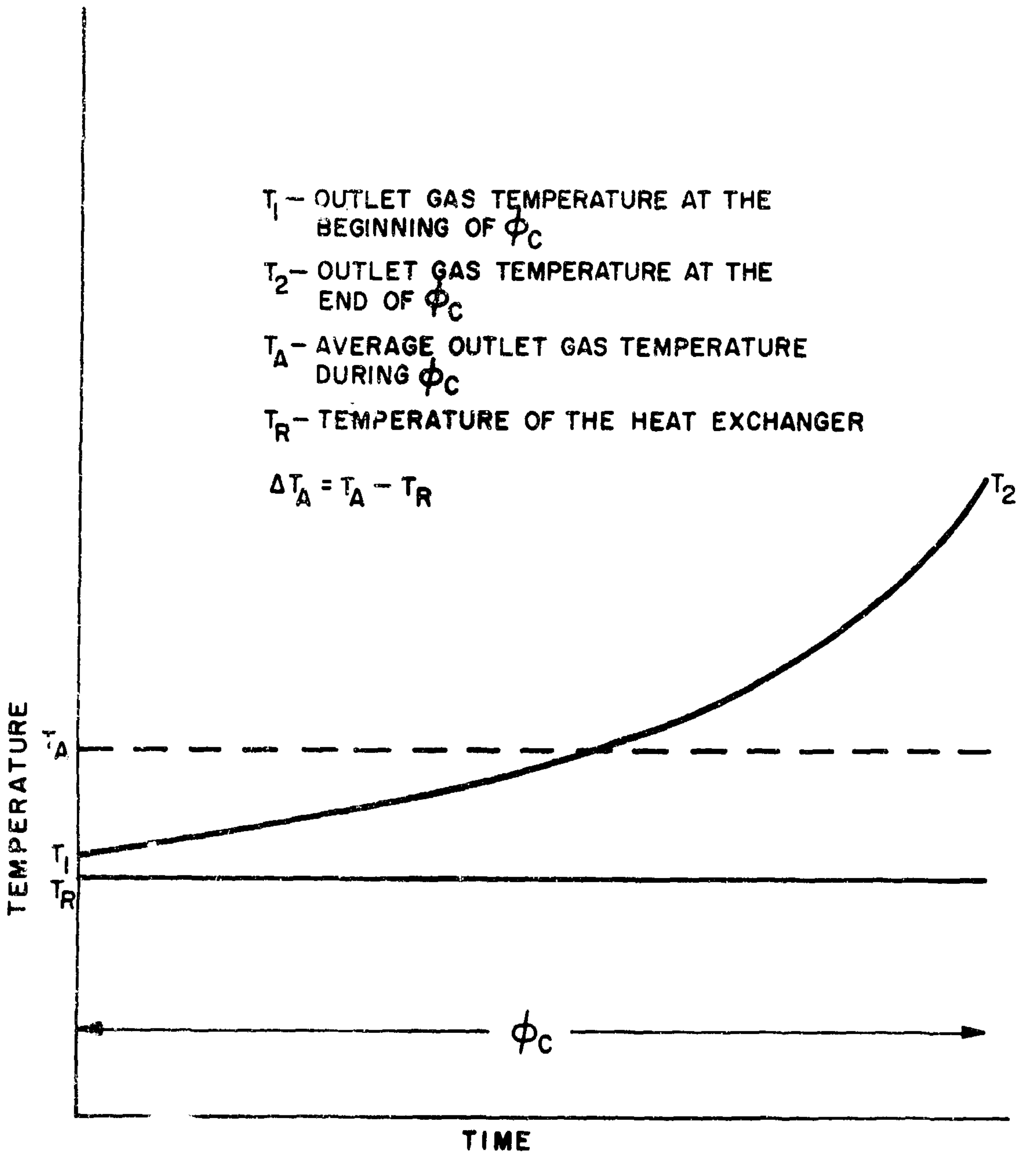

FIG.3 REGENERATOR TEMPERATURE VARIATION
(GAS TEMPERATURE AT COLD END)

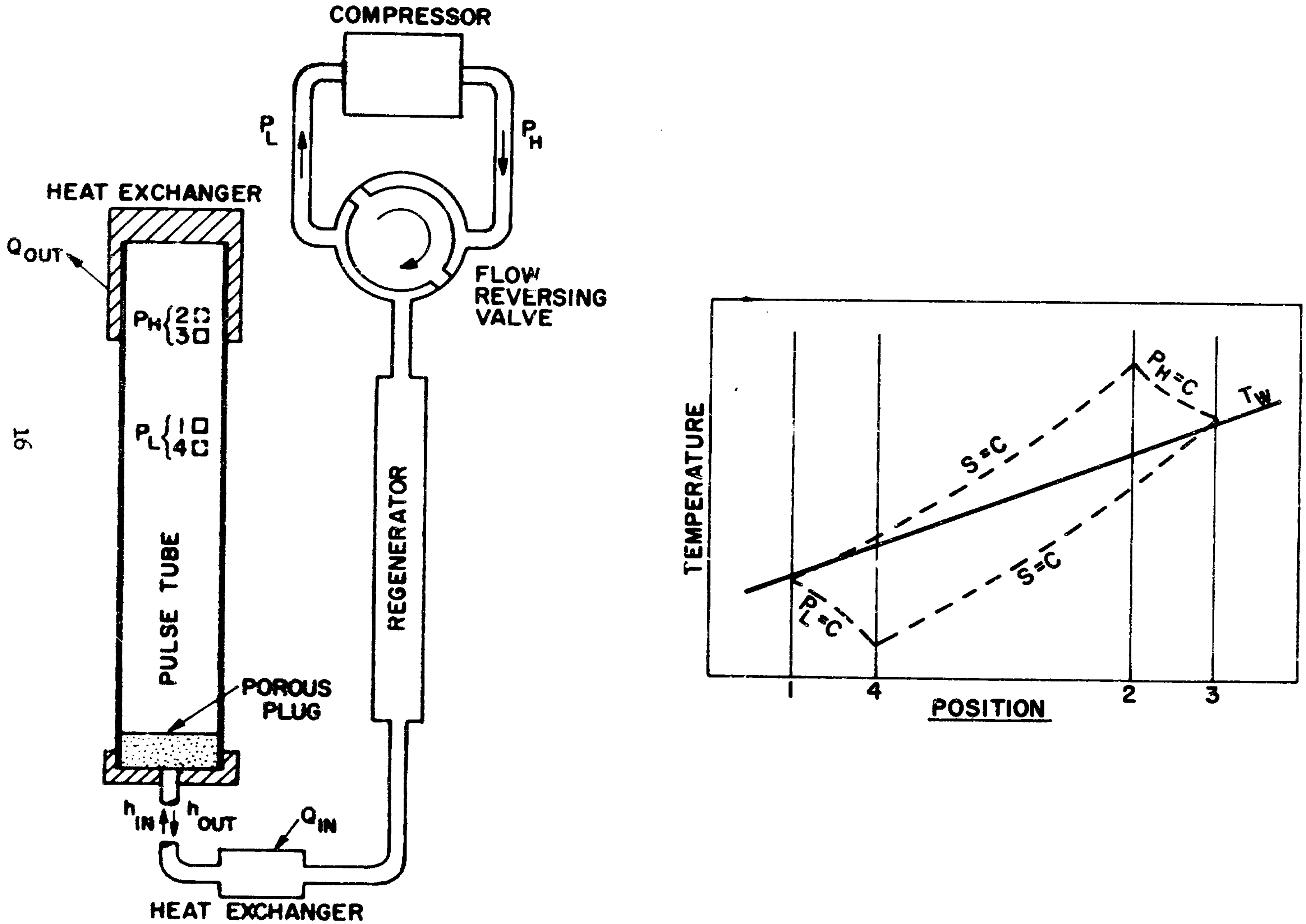

FIG.4 PULSE TUBE CYCLE

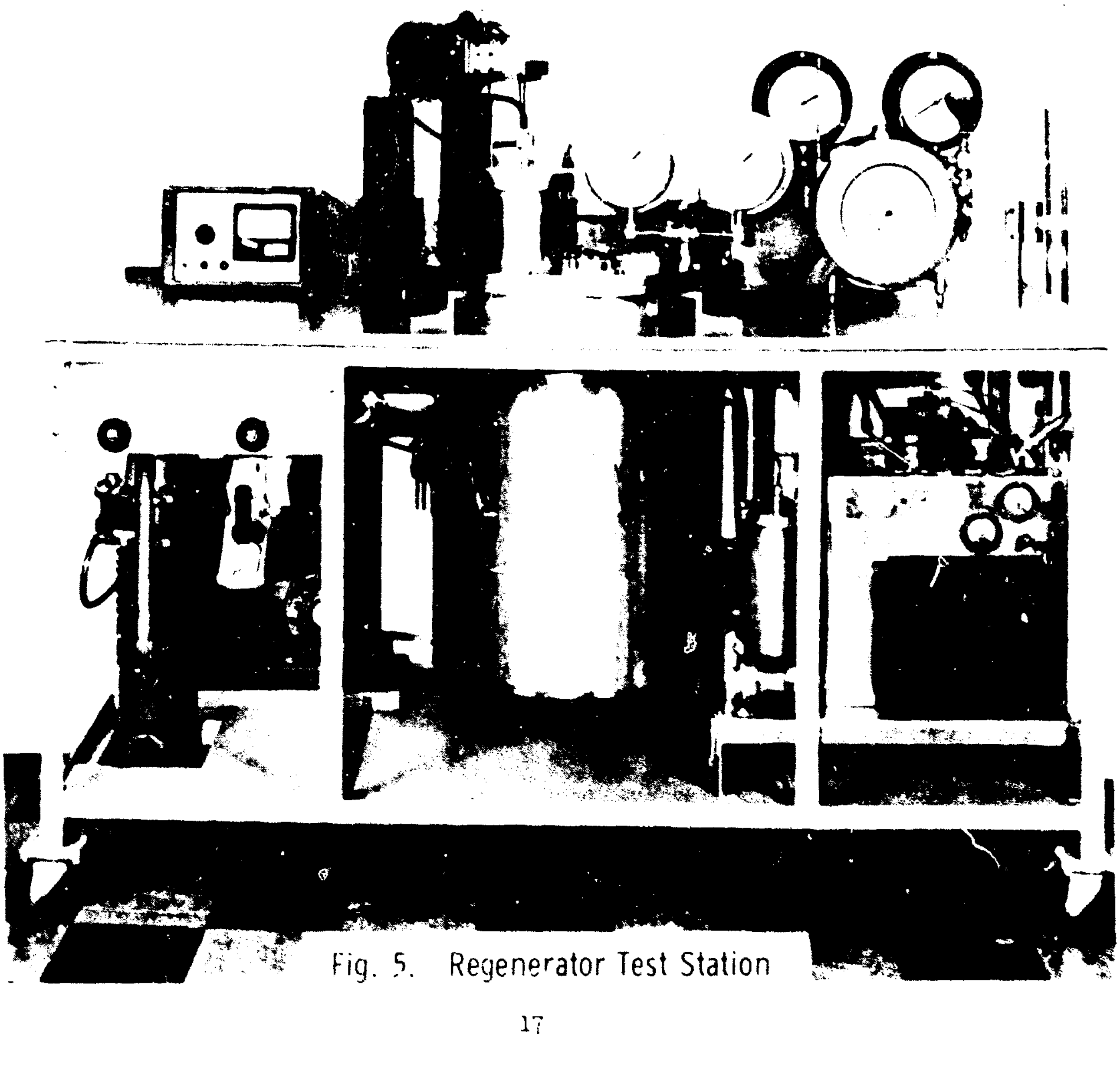

Fig. 5. Regenerator Test Station

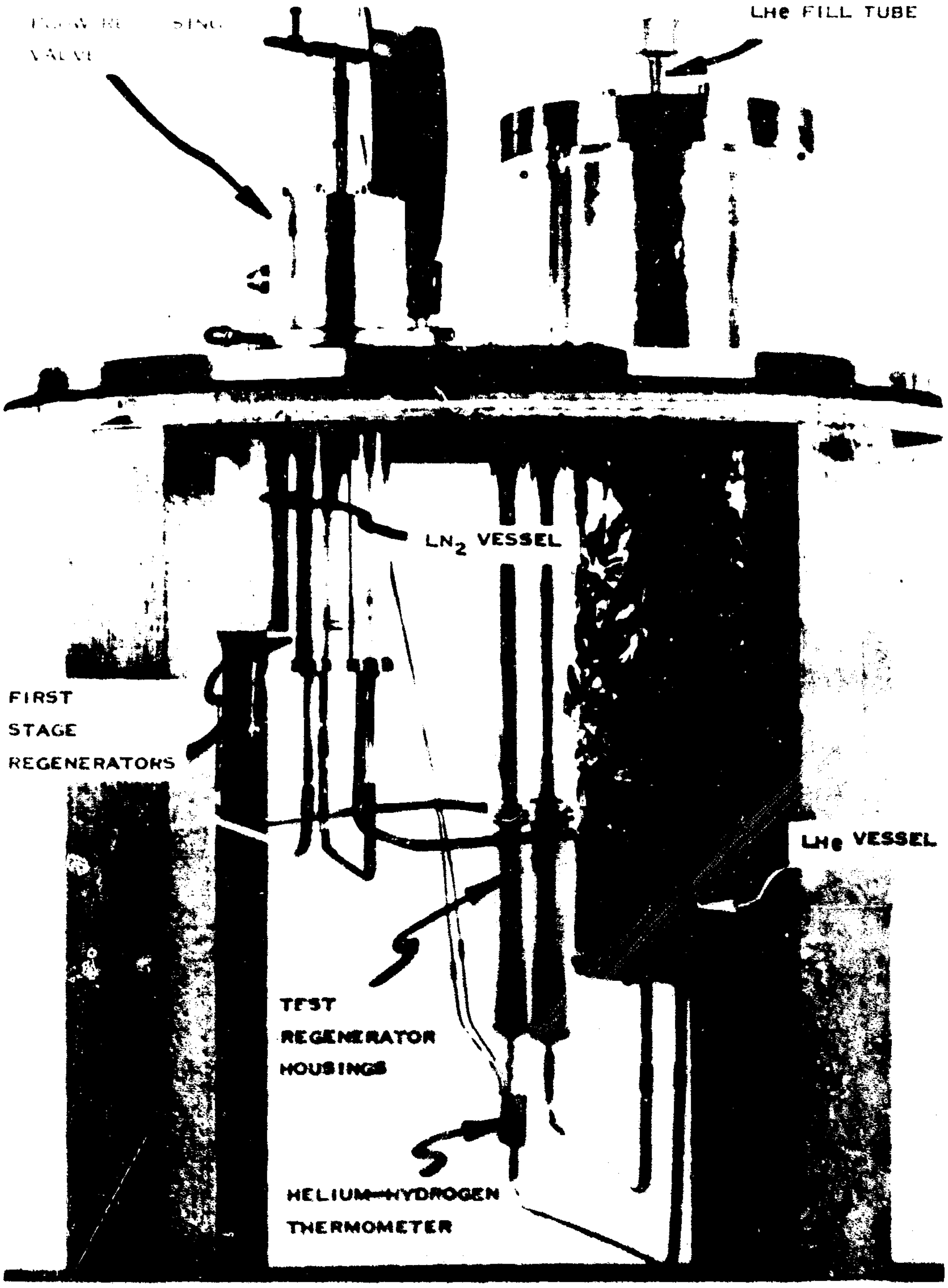

Fig. 6. Internal View of Regenerator Test Station

REGENERATOR TEST STATION

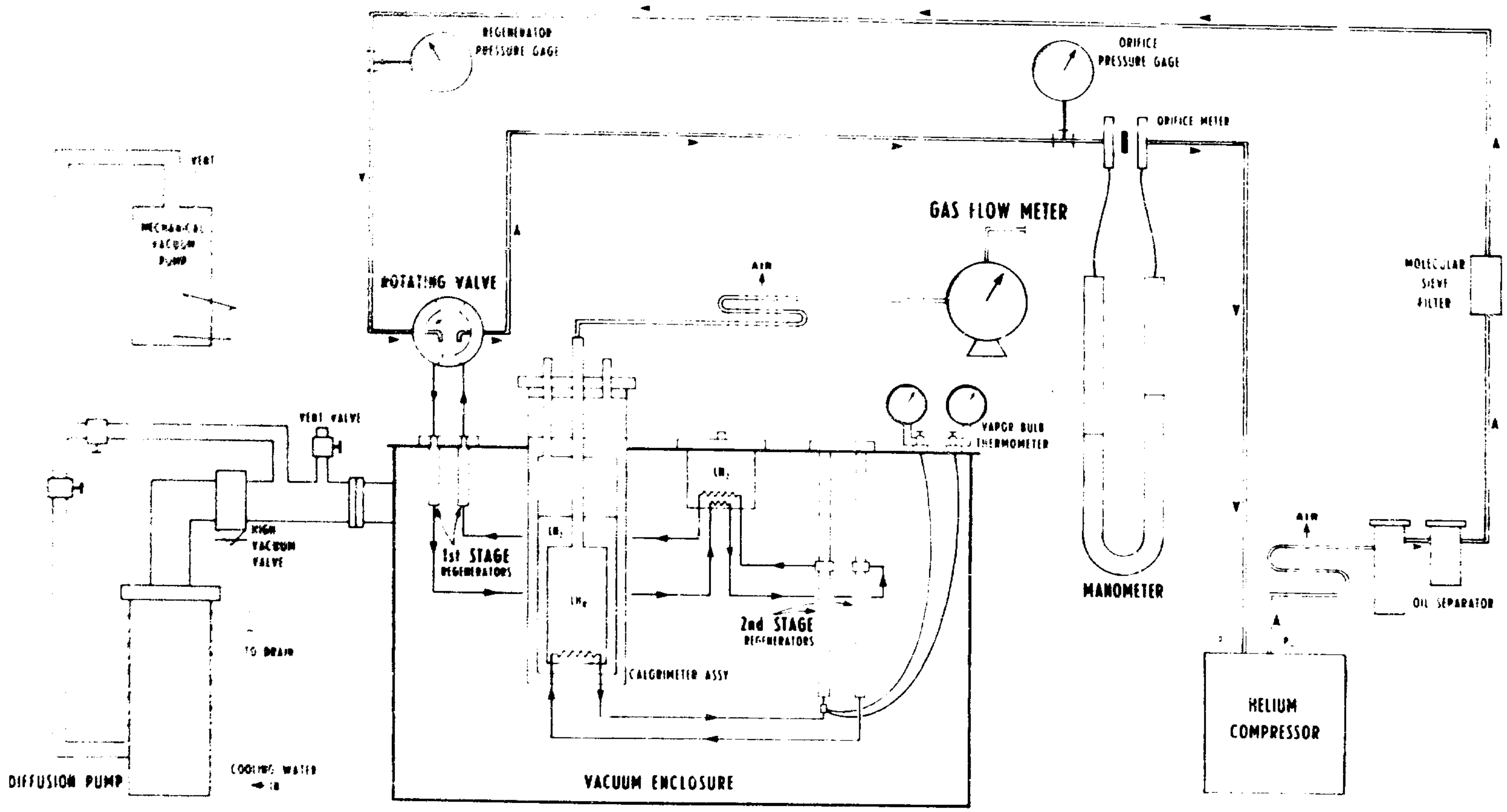

Fig. 7. Regenerator Test Station Diagram

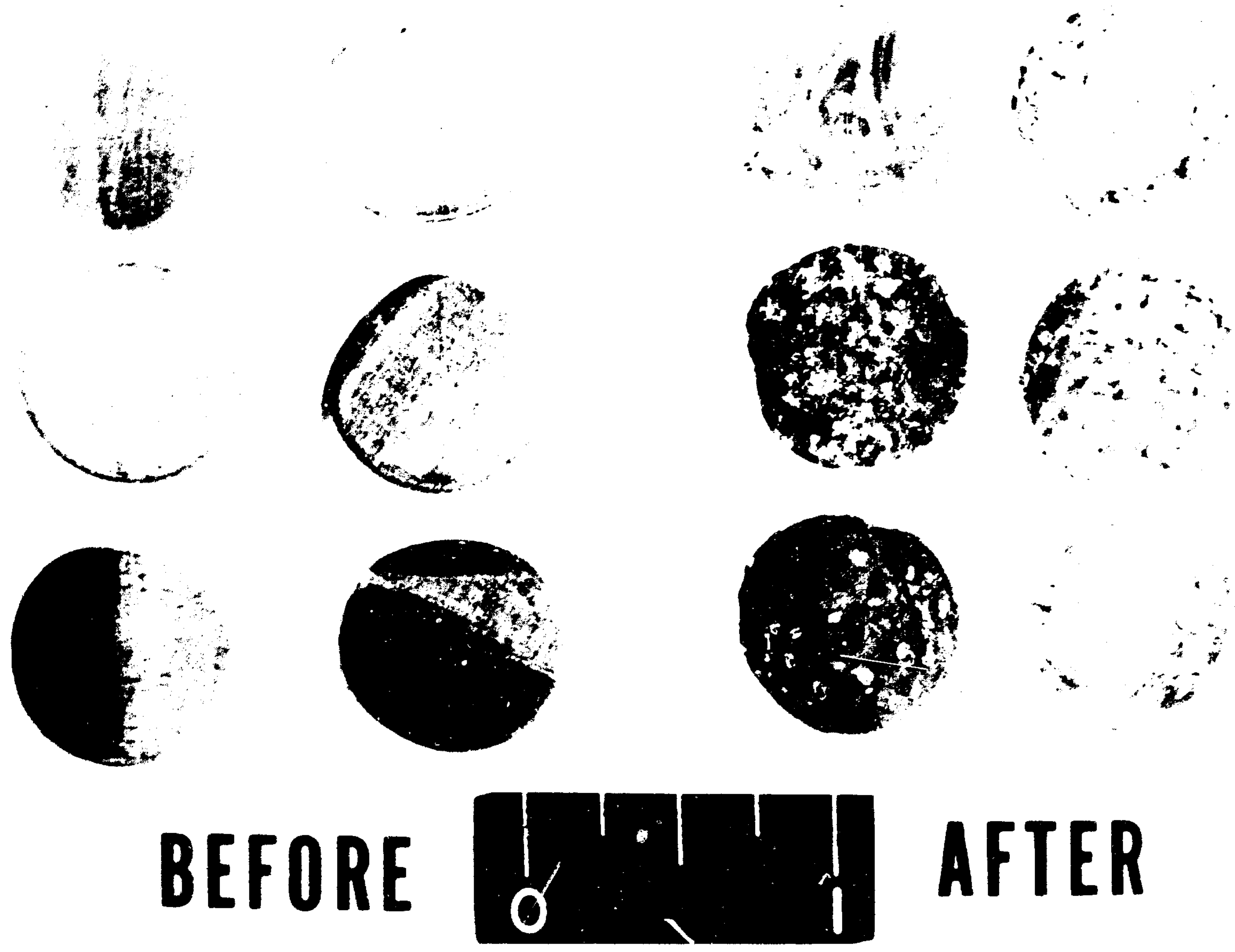

Fig. 8. Corrosion of Regenerator Screens

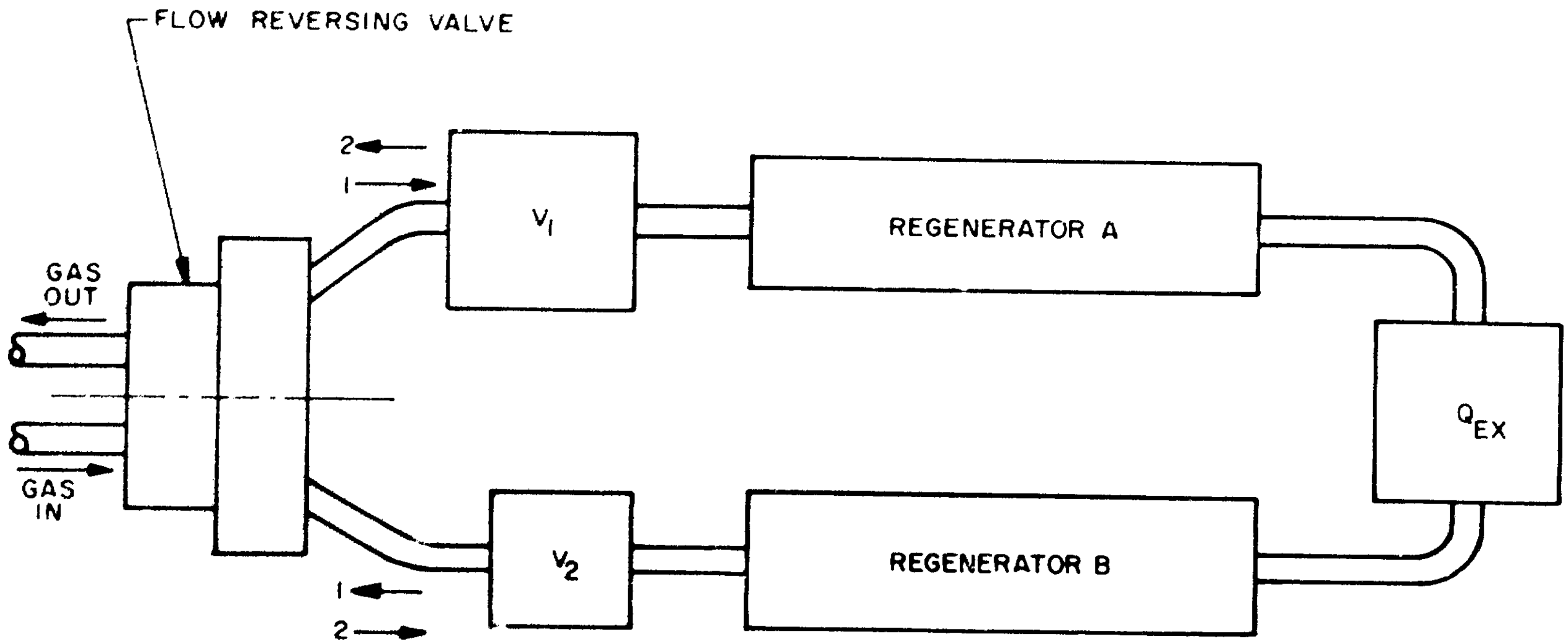

FIG. 9 UNBALANCED FLOW DIAGRAM

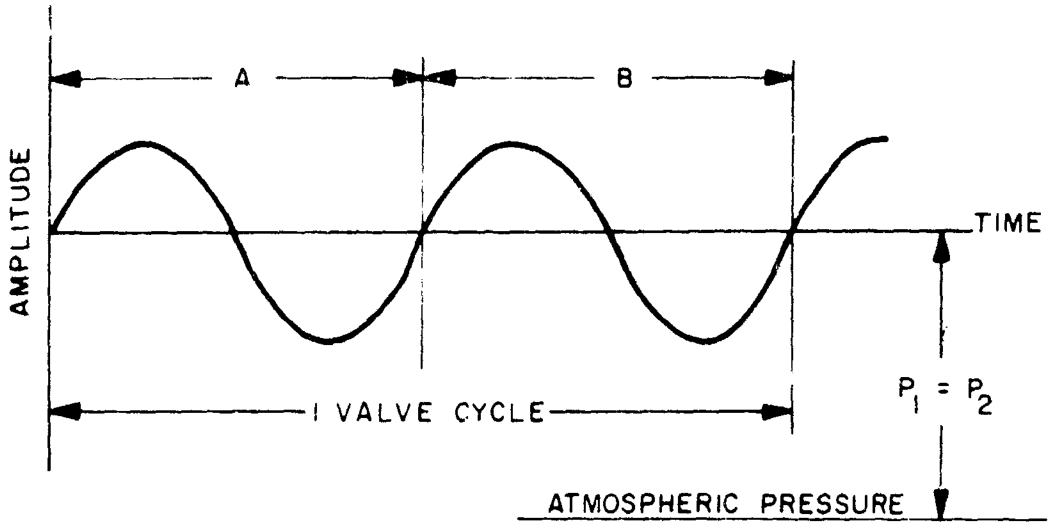

FIG. 10 BALANCED GAS FLOW PRESSURE VARIATION

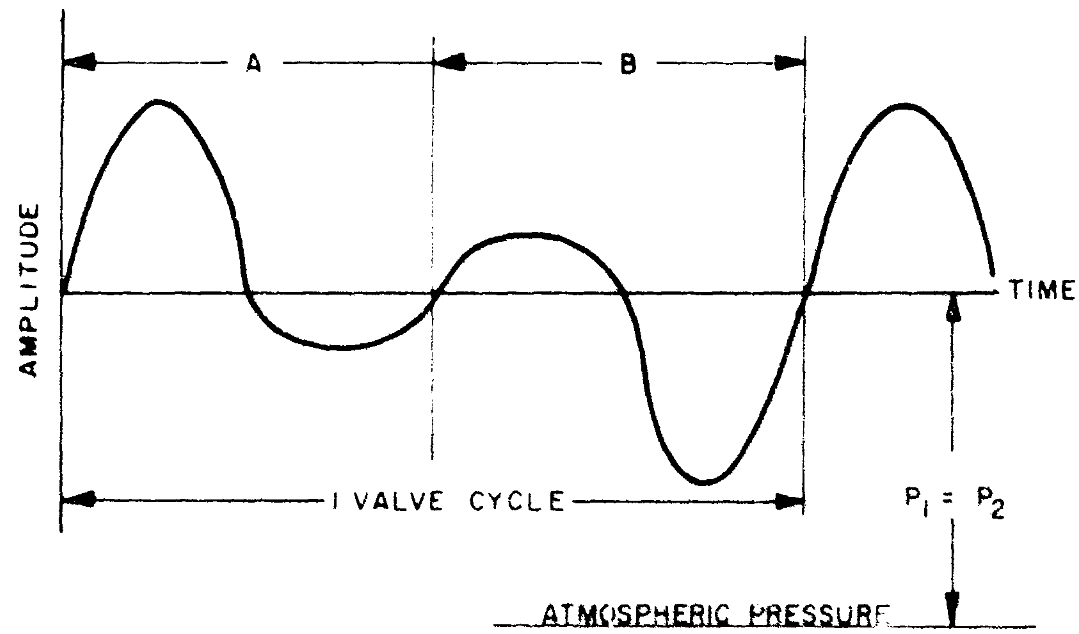

FIG. 11 UNBALANCED GAS FLOW PRESSURE VARIATION

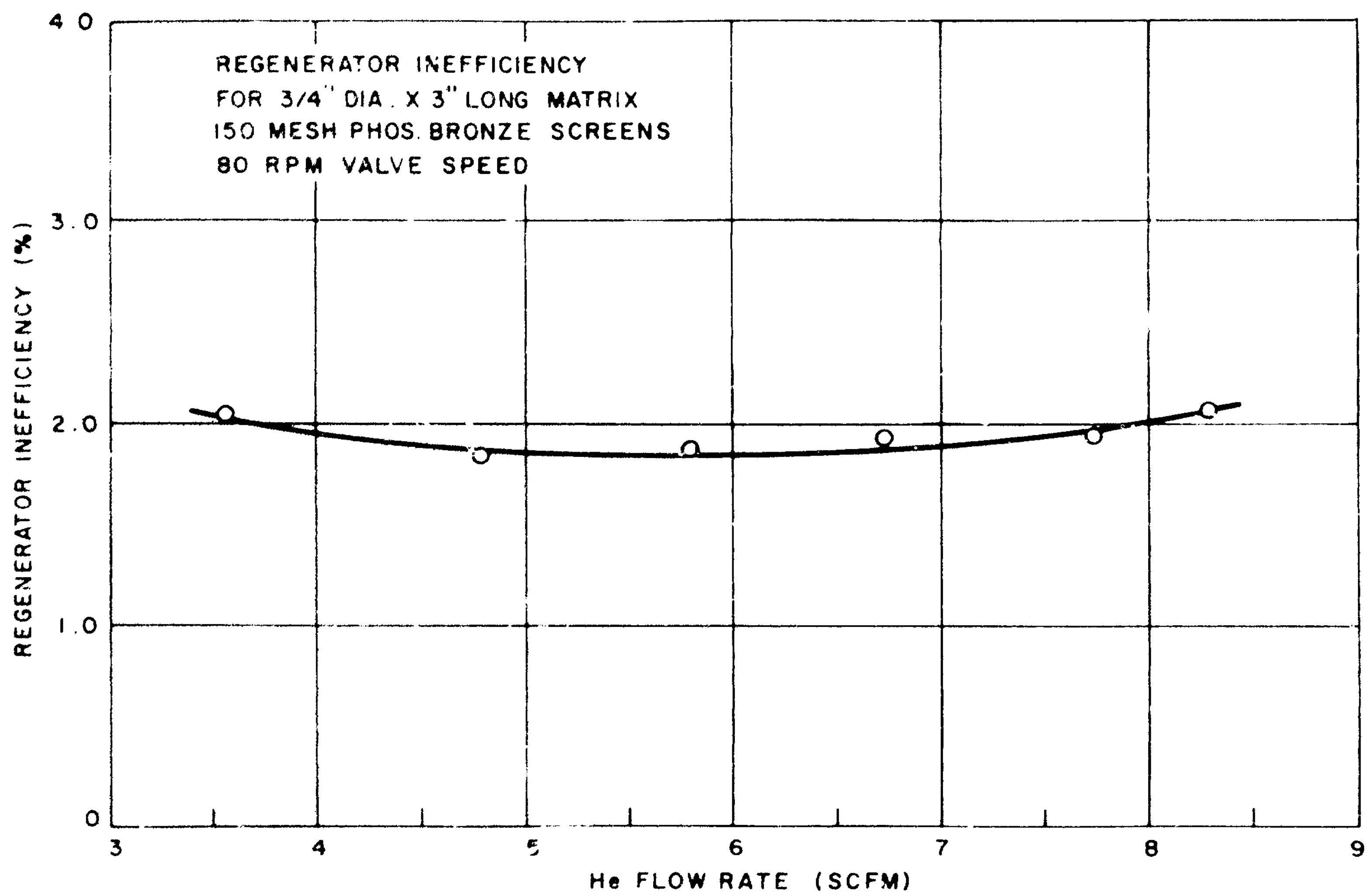

FIG. 12 REGENERATOR INEFFICIENCY VS. FLOW RATE

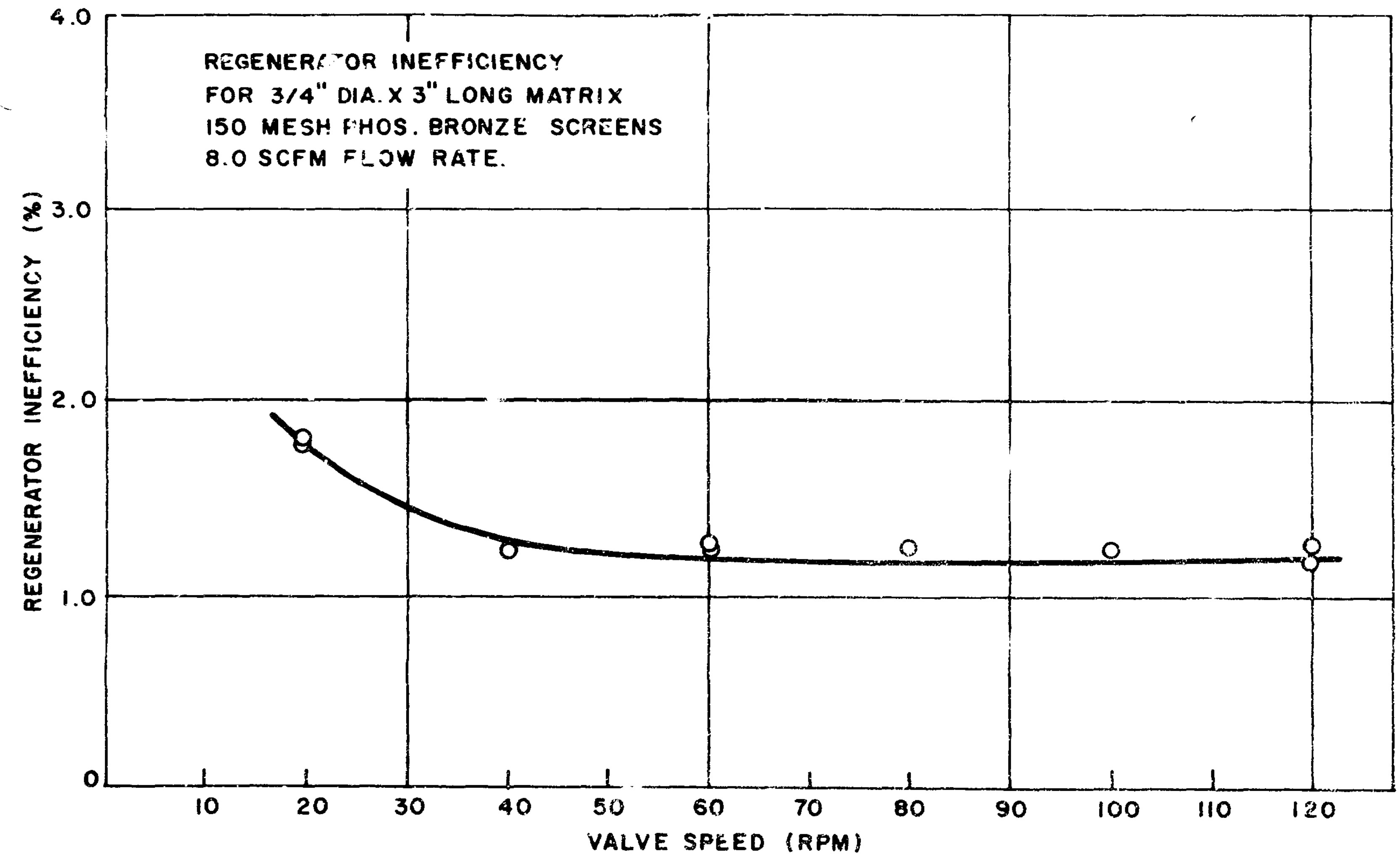

FIG. 13 REGENERATOR INEFFICIENCY VS. VALVE SPEED

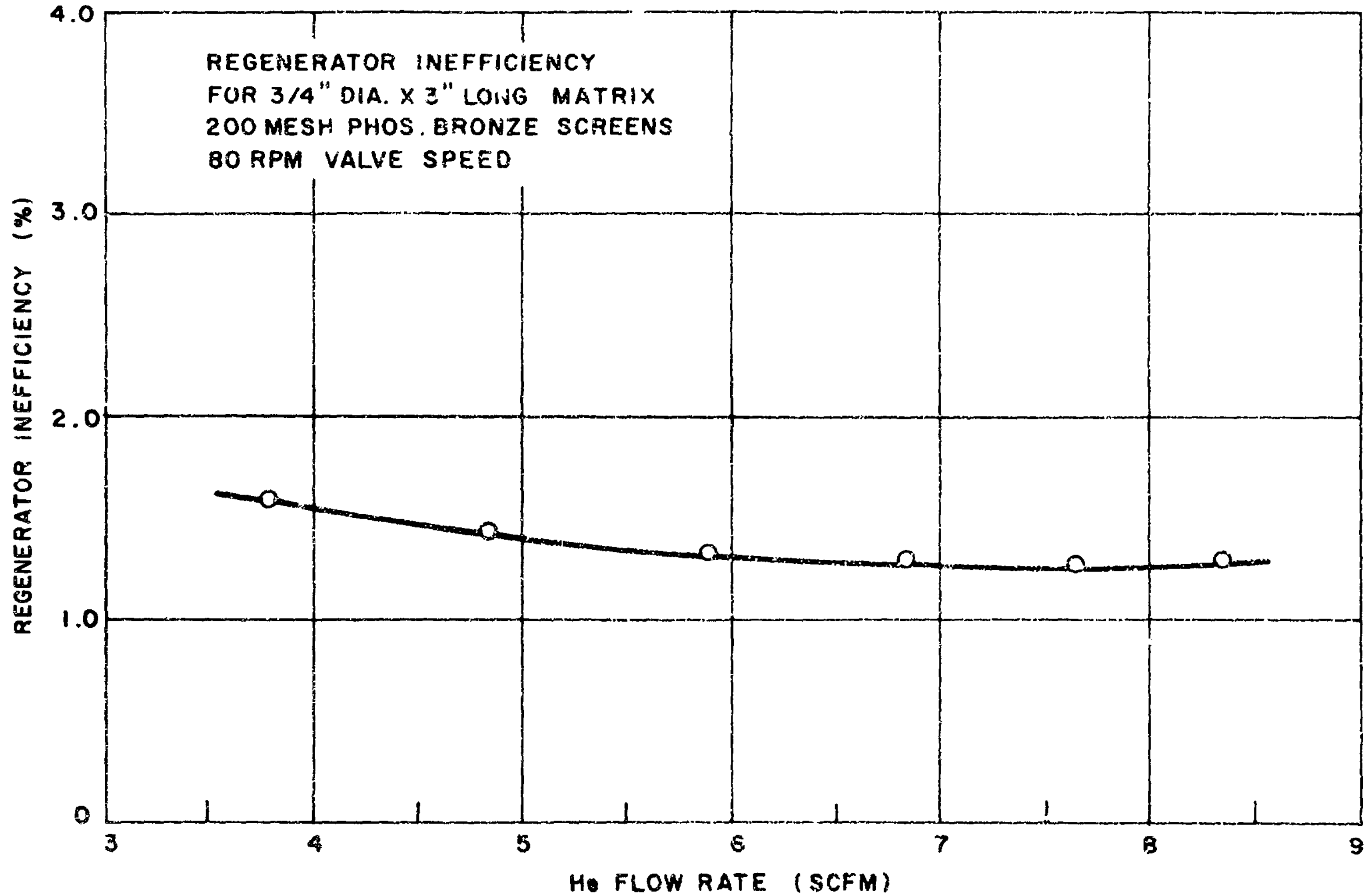

FIG. 14 REGENERATOR INEFFICIENCY VS. FLOW RATE

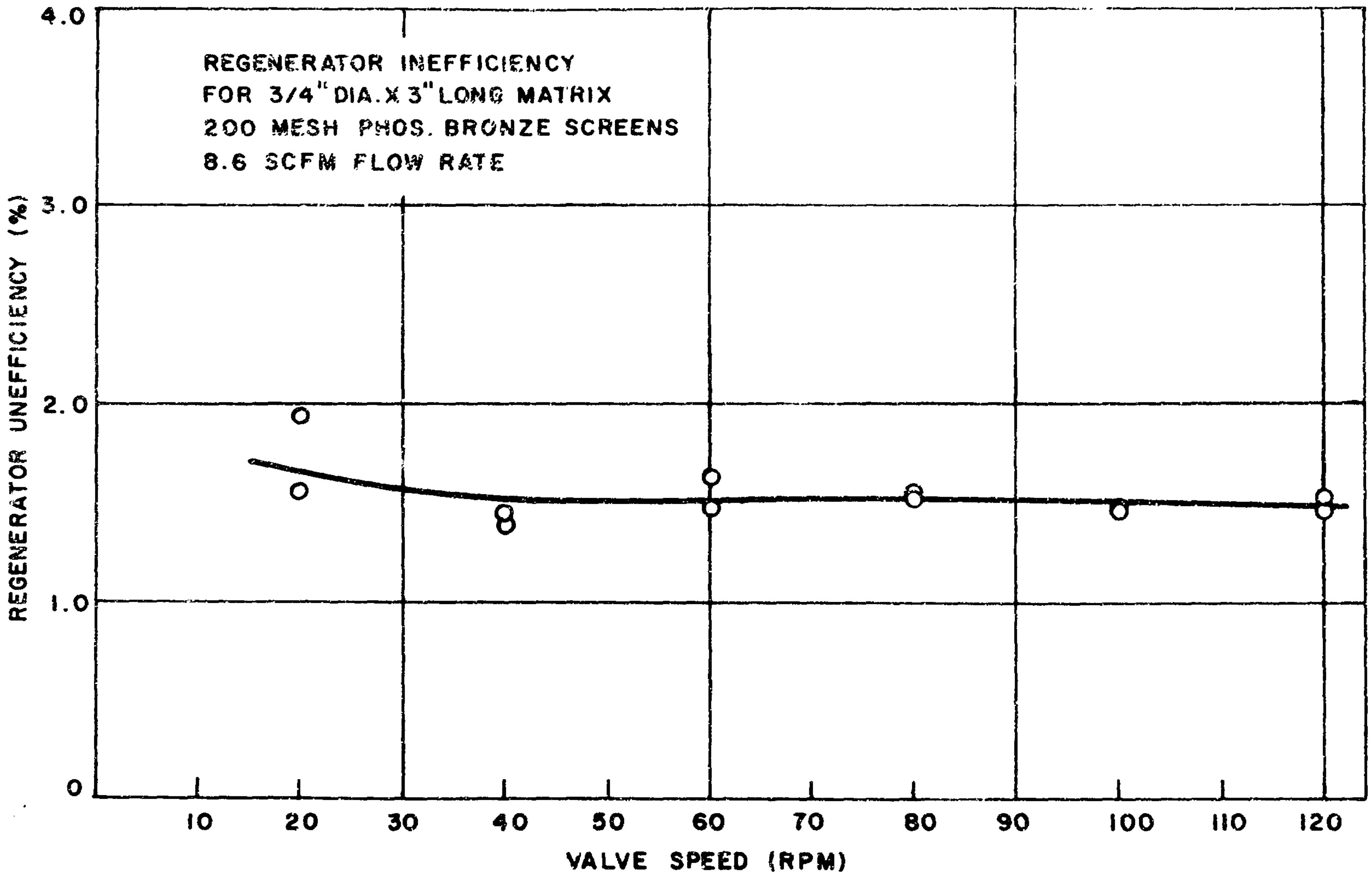

FIG. 15 REGENERATOR INEFFICIENCY VS. VALVE SPEED

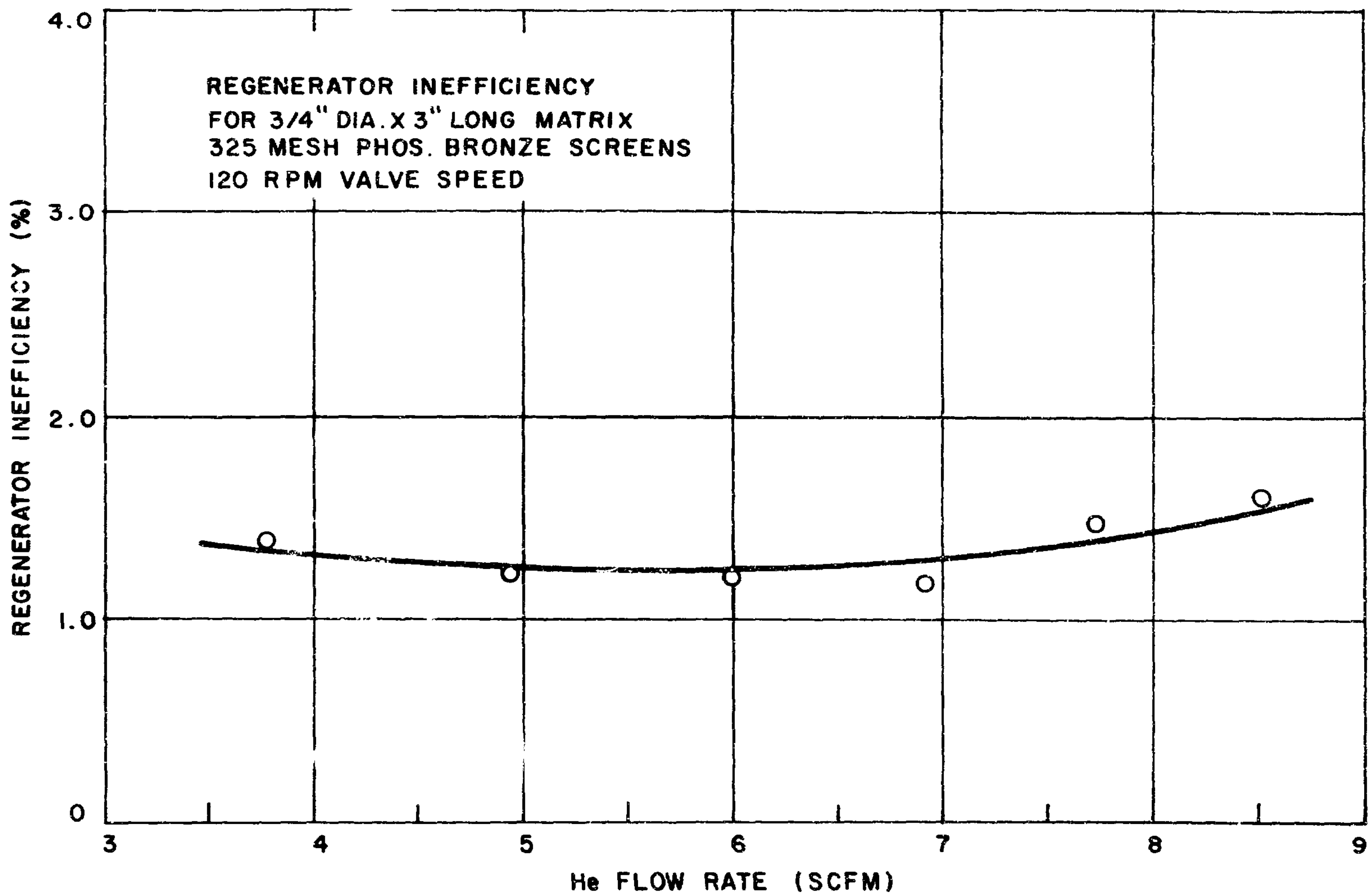

FIG. 16 REGENERATOR INEFFICIENCY VS. FLOW RATE

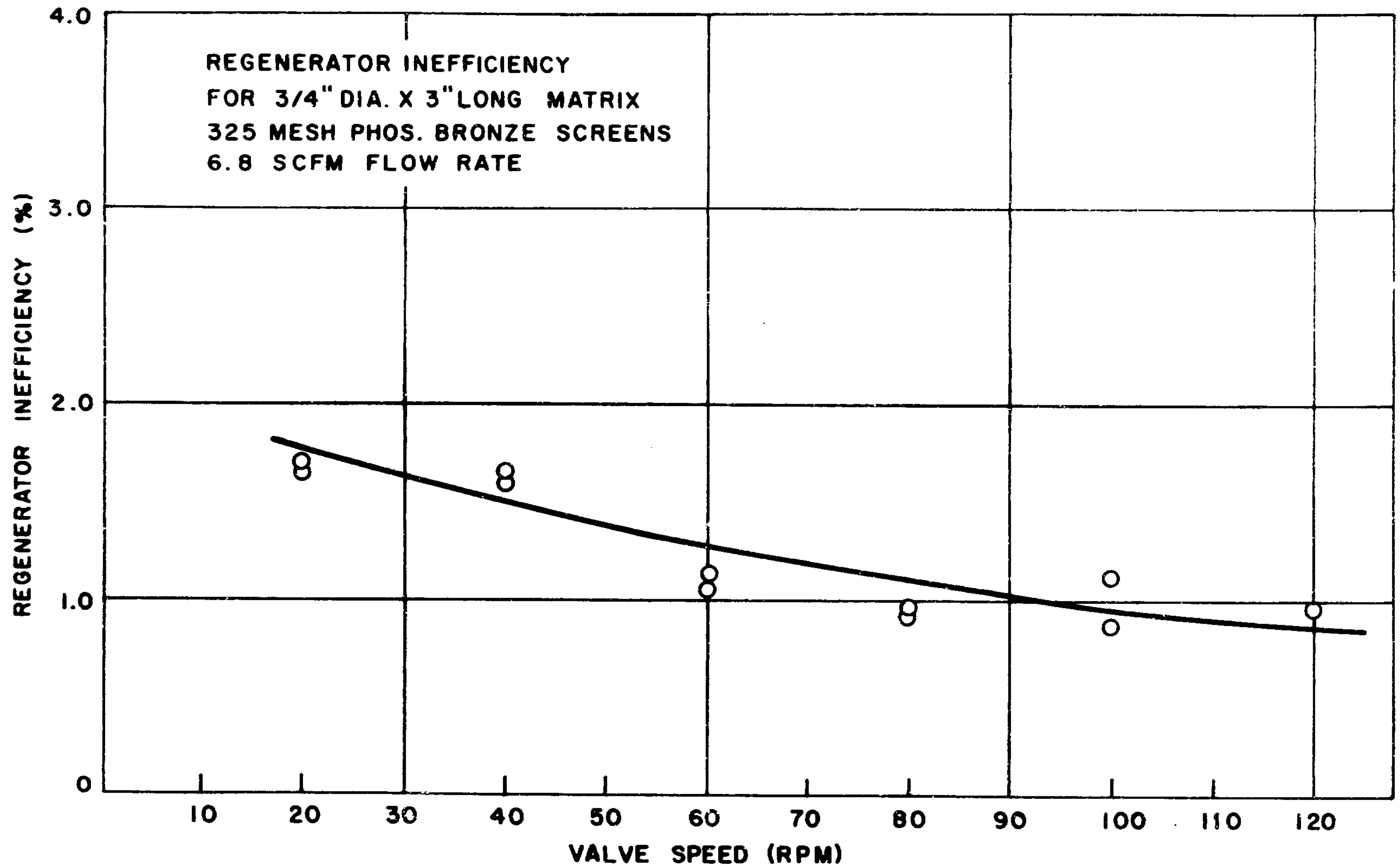

FIG 17 REGENERATOR INEFFICIENCY VS. VALVE SPEED

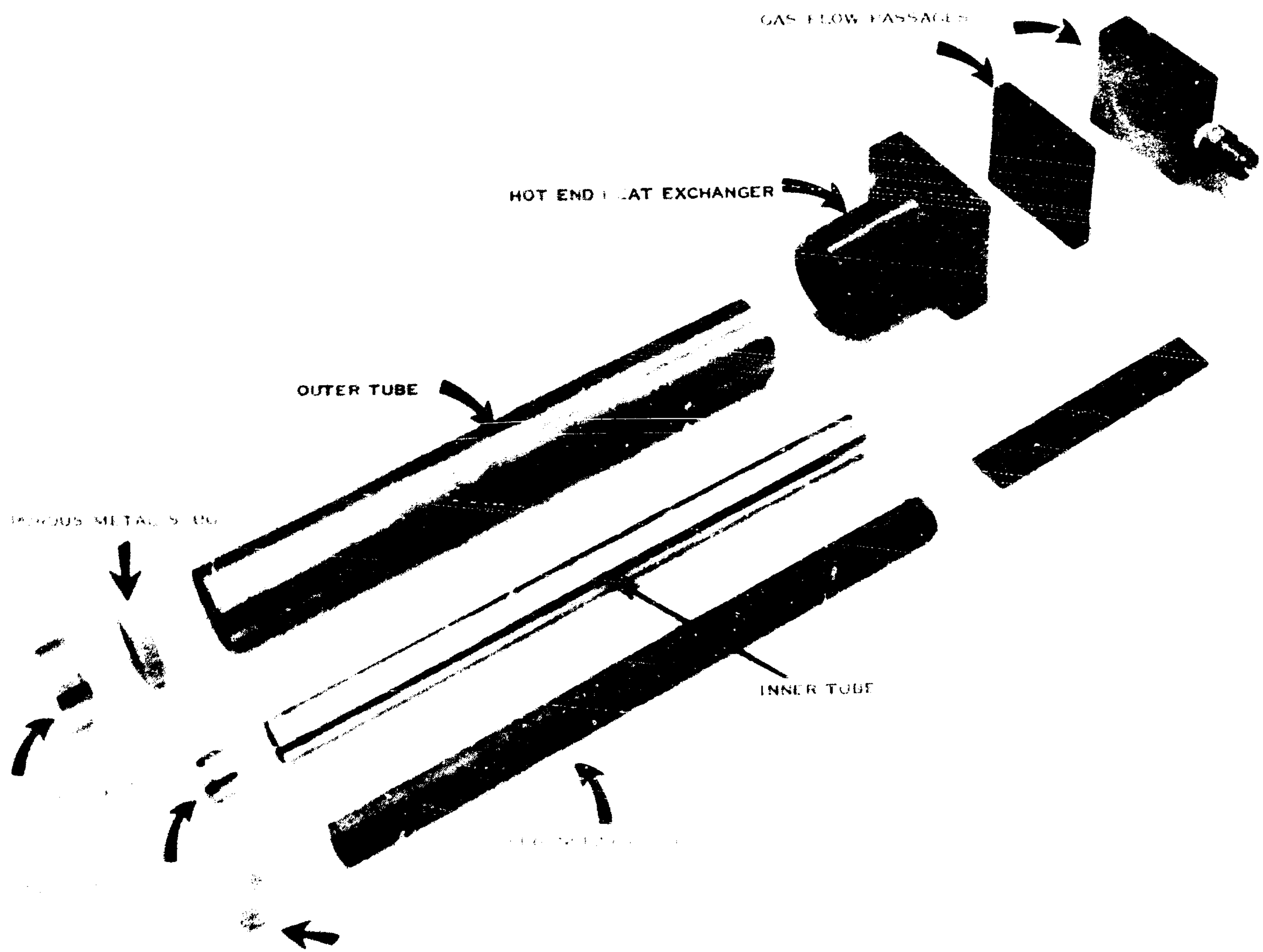

Fig. 18. Concentric Pulse Tube Refrigerator (Pre-Assembly)

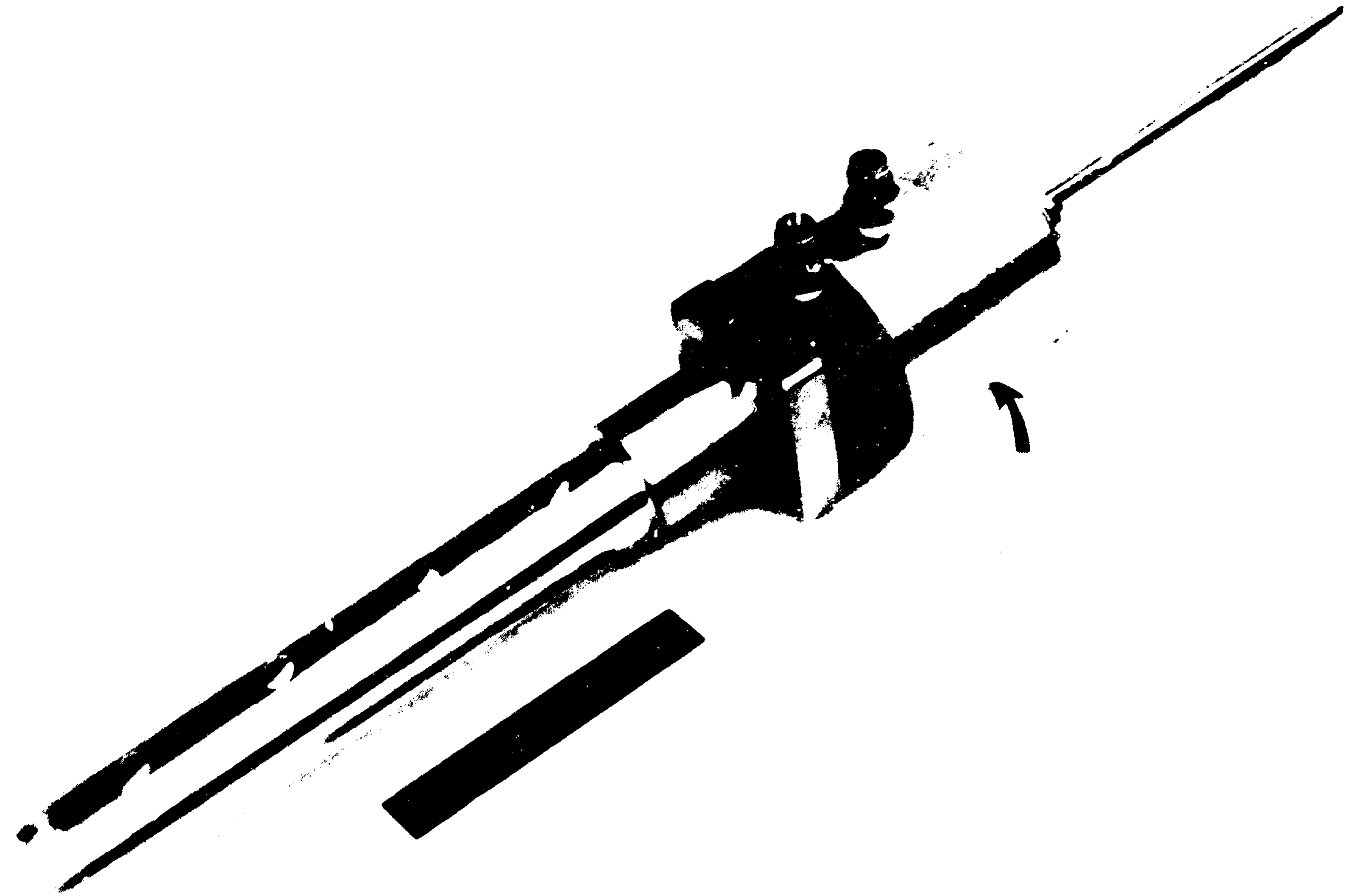

Fig. 19. Concentric Pulse Tube Refrigerator (Assembly)

APPENDIX I

REGENERATOR TEST COMPUTER PROGRAM

```
READ/,TR,PO,PM,BL,BO
TR=TR+460
PO=PO+14.7
PM=PM*5.2
F=0.0353
CD=0.607264
A=0.0001199
R=386.0
HV=85.67
C=60*CD*A
D=PO*144/(TR*R)
CFM=C*(64.4*PM/D)**0.5
RATE=CFM*D
QT=RATE*1.25*(TR-139)
DN=2116.8/(TR*55.1)
QL=HV*DN*F*(BO-BL)
XEFF=(QL/QT)*100
SCFM=PO*520*CFM/(14.7*TR)
PRINT 2,C,D,CFM,QT,DN,QL,XEFF,SCFM
FORMAT (4F10.6,/,4F10.6)
```